START

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August 1, 1985 65610-85-186

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Expected Drill Bit Temperature During Sample Taking in Single Shell Tanks

Introduction

The waste in several single-shell tanks are to be characterized. This characterization requires samples of the waste be obtained. The samples are planned to be taken by using a drilling rig to take 20 inch sample cores. A determination of the expected drill bit temperature was needed for the Safety Analysis Report (SAR). This work addresses some temperature questions that occurred in the development of the SAR for the waste tank core sampling (to be released as SD-WM-SAR-007).

Summary

Sampling of waste in single-shell tanks is to be done using the tank core sampling equipment. A core drilling rig will drill into the waste to retrieve samples. The bit temperature is important because some of the tanks contain ferrocyanide that could undergo an exothermic reaction if temperatures significantly exceed 300°C (572°F). Heat generation rates were calculated as a function of applied hydraulic pressure, which is used to increase the downward force on the drill bit, drill rpm, and the friction factor. This heat generation rate was used in a HEATING5 heat transfer model to determine the expected heat rise. Assuming a sliding friction factor, the maximum hydraulic pressure, and 500 rpm, the drill bit temperature will not increase more than 103°C. This temperature rise with the latest tank temperature data show that no tanks will exceed the 300°C (572°F) limit (See Tables 1 and 2).

Discussion

To calculate the temperature of the drill bit requires that the heat generated by the action of the bit be determined. It was assumed that all of the energy needed to turn the drill was converted to heat. In Appendix A is an equation for calculating the heat generated. Heat generated by the drill bit is related to the friction factor, the hydraulic pressure applied to the bit, and the rpm of the drill.

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The analysis assumed values of friction factor for a worst case of running the drill into the bottom of the tank. This condition was assumed because the friction factor is expected to be higher for steel on steel as opposed to steel on waste. The friction factor used for the drill turning and then touching the bottom was .12. This friction factor is for sliding greasy conditions. Greasy conditions apply when two rubbing surfaces are separated by a very thin film of lubricant. This kind of condition would be expected to exist because of the tanks are not totally dry and the bit will be turning when it touches the bottom of the tank. For dry (non greasy) conditions, the sliding friction factor would be .41.

For the case of having the drill bit sitting on the bottom and then starting to spin the drill, the friction factor is .23. For dry conditions, the static friction factor would .78. The static friction factors are for static conditions which would not apply after the drill starts to move. This condition would only exist for a few seconds after torque was applied to the drill string.

The hydraulic pressure is used to create additional downward force on the drill bit. The pressure is applied directly to rams which in turn create a downward thrust on the bit. A maximum of 2000 psi can be applied to the bit by the way of two 2-inch rams. The drill bit temperature was calculated for various applied hydraulic pressures from 0 to 2000 psi. Two drill speeds were considered: 500 rpm and 200 rpm. The analysis assumed that the drill would be running against the bottom for between 1 and 6 minutes (this is expected to be conservative especially for the cases using static friction factor since static conditions only exist for a few seconds.)

The various heat generation rates calculated were used in a HEATING5 heat transfer model to determine temperature increase as a function of the heat generated by the bit. The assumptions and simplifications used for the HEATING5 heat transfer model are outlined in Appendix B. The heat transfer up the drill string was included in the heat transfer model. The model did not include the effect of air or fluid that will be fed down the drill string.

The temperature rise versus heat generation rate was used to determine the temperature of the drill bit. Calculations of the heat generation and the results of temperature rise are shown in Appendix A.

Results

Using various applied hydraulic pressures and sliding friction factors, the bit temperature increase can vary from 0 to 103°C (0 to 173°F), see Table A.1 and A.2 in Appendix A. Table 1 shows the expected bit temperatures for all the tanks to be sampled assuming that the full hydraulic pressure (2000 psi) is applied to the drill bit, the drill was turning at 500 rpm for 6 minutes and the friction factor is for sliding conditions - greasy and dry. Under greasy conditions, the drill bit temperature would rise by 30°C. For the hottest tank, the temperature of the drill bit would reach 100°C. If the single-shell tanks are totally dry, the temperature rise would be 103°C, causing the bit to reach 173°C for the hottest tank. Table 2 is for 300 psi applied hydraulic pressure, 200 rpm and drill spinning of one minute with friction factors for sliding conditions. The temperature rises are 1 and 3°C for greasy and dry situations, respectively.

Conclusion

The results of the calculations show that drill bit temperatures will not cause ferrocyanide to undergo an exothermic reaction. In the worst case of trying to drill through the bottom of the tank and applying full pressure, the drill bit temperature will be 100 and 173°C for greasy and dry sliding conditions, respectively. These temperatures are well below the 300°C temperature limit.

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DCR/sjw

Att.

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TABLE 1

Tank Number	Maximum Tank Temperature °F	°C	Bit Temp, °C 2000 psi Applied 300 rpm for 6 min. Sliding Condition Greasy	Dry	Date of Measurement
Friction Factor			.12	.41	
BY-104	158	70	100	173	3/03/85
BY-105	133	56	86	159	4/01/85
BY-106	140	60	90	163	4/01/85
BY-107	90	32	62	136	10/07/78
BY-108	99	37	67	141	3/05/85
BY-110	140	60	90	163	5/17/85
BY-112	93	34	64	137	6/21/81
C-101	82.9	28	58	132	1/15/85
C-108	83	28	58	132	8/05/83
C-109	87	31	61	134	8/05/83
C-111	83	28	58	132	8/05/83
C-112	79	26	56	129	8/05/83
T-101	72	22	52	126	5/10/83
TY-101	68	20	50	123	11/13/83
TY-102	65	18	48	122	11/13/83
TY-103	76	24	54	128	2/07/81
TY-104	68	20	50	123	11/13/83
TY-105	89	32	62	135	11/13/83
TY-106	68	20	50	123	11/13/83

TABLE 2

Tank Number	Maximum Tank Temperature °F	°C	Bit Temp, °C 300 psi Applied & 200 rpm Sliding Condition for 1 Minute Greasy	Dry	Date of Measurement
Friction Factor			.12	.41	
BY-104	158	70	71	73	3/03/85
BY-105	133	56	55	59	4/01/85
BY-106	140	60	61	63	4/01/85
BY-107	90	32	33	35	10/07/78
BY-108	99	37	38	40	3/05/85
BY-110	140	60	61	63	5/17/85
BY-112	93	34	35	37	6/21/81
C-101	82.9	28	29	31	1/15/85
C-108	83	28	29	31	8/05/83
C-109	87	31	32	34	8/05/83
C-111	83	28	29	31	8/05/83
C-112	79	26	27	29	8/05/83
T-101	72	22	23	25	5/10/83
TY-101	68	20	21	23	11/13/83
TY-102	65	18	19	21	11/13/83
TY-103	76	24	25	27	2/07/81
TY-104	68	20	21	23	11/13/83
TY-105	89	32	33	35	11/13/83
TY-106	68	20	21	23	11/13/83

APPENDIX A

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DESIGN ANALYSIS

PAGE 1.4
JOB NO.
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BY [illegible]
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FOR
LOCATION
SUBJECT [illegible]

Moment to Turn Bit

$$M = \tfrac{2}{3}\,\mu P \left[\frac{R_o^3 - R_i^3}{R_o^2 - R_i^2}\right]$$

from [illegible]
2nd edition, pg [illegible]
[illegible]

where M = moment, [illegible]

[illegible]

[illegible]

[illegible]

[illegible]

[illegible]

[illegible]

[illegible]

[illegible]

[illegible]

[illegible]

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DESIGN ANALYSIS

PAGE 2/4
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BY D.C. [illegible]
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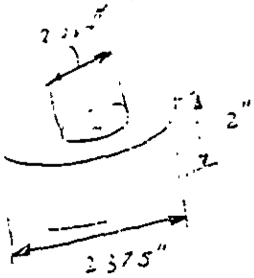
FOR
LOCATION
SUBJECT Drill 3 - [illegible]

Convert [illegible]

$$\frac{[\ldots\ 22\ \text{ft-lb}\ \ldots\text{min}][60\ \text{min/hr}]}{778\ \text{ft-lbs/Btu}} = \ldots\ \text{Btu/hr}$$

Volume that the drill heats up [illegible]

Assume that the most [illegible] the bit



$$\text{Volume} = \pi \frac{\left[\left(\frac{2375}{2}\right)^2 - \ldots\right]\ldots}{\ldots\ \text{in}^3/\ldots}$$

Volume = .00[illegible] ft³

Heat [illegible] Rate

$$\frac{\ldots\ \text{Btu/hr}}{.00[illegible]\ \text{ft}^3} = \ldots\ \text{Btu/hr-ft}^3$$

Pressure on the Bit

$P = P_{\text{applied}} + P_{\ldots}$

Pressure applied by [illegible]

The pressure applied to the 2" [illegible]

$P_{\text{applied}} = [\text{Pressure} \ldots][2\pi \ldots] \ldots$

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PAGE 3/4
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FOR
LOCATION
SUBJECT Drill [illegible]

<u>Pressure applied by weight of drill string</u>

P drill string = Weight of drill string and sampler [illegible]

Length of Drill string

Distance from ground to [illegible]
Distance from ground to bed of truck = [illegible]
Length of barrel = 3.75 feet
57.75 [illegible]

Pipe drill string is made of 2" nominal schedule [illegible] pipe [illegible]
weight of drill string/ft = 3.5[illegible] lb/ft

Length of sampler = [illegible] inches

Sampler made of pipe : 1½ nominal [illegible] Schedule [illegible] } [illegible]

Weight of sampler [illegible]
I.D. of sampler = 1.77[illegible]
[illegible]
[illegible] 1.77[illegible]

Weight of sample/ft = $\left[\frac{\pi(1.77 \ldots)^2}{(4)(144\ \mathrm{in}^2/\mathrm{ft}^2)}\right]$ ([illegible]) = 2.122 lb/ft

Drill String weight = (57.75 ft)(3.[illegible]2 lb/ft) = [illegible]

Sampler and sample weight = $\left(\frac{[\text{illegible}]\ \text{inches}}{12\ \text{in/ft}}\right)$(1.2[illegible] + 2.122 lb/ft) = 5.[illegible] lbs
222.[illegible] lbs

<u>Test pressure</u>

P = [illegible] [Pressure applied] = [illegible]

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DESIGN ANALYSIS

PAGE 4/4
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FOR
LOCATION
SUBJECT Drill Bit Heat-up

Friction Factor

The bottom of the tank may be [illegible] [illegible] greasy friction factors were applied. The following friction factors came from

Marks' Mechanical Engineers Handbook
Theodore Baumeister, Ed. [illegible] 3-[illegible]
(see attached sheet)

Friction Factors:

Condition:	Factor
Sliding Greasy	.12
Sliding Dry	.42
Static Greasy	.33
Static Dry	.79

Heat Generation Rate

heat generation rate = 3.[illegible] [f] [6.3 (Pressure) − 222.7] $\frac{Btu}{hr\text{-}ft^3}$

f can vary from .12 to .79

Pressure can vary from [illegible] to [illegible]300 psi

STATICS

SECOND EDITION

J. L. Meriam

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SCHOOL OF ENGINEERING
DUKE UNIVERSITY

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NEW YORK LONDON SYDNEY TORONTO

clearance space with a change from zero at the fixed inner surface of the bearing to the peripheral velocity v of the shaft at its outer surface. For the radial clearance c, the velocity gradient has the magnitude $|dv/dr| = v/c = r\omega/c$, where ω is the angular velocity of the shaft in radians per second. The shear stress on the surface of the shaft from Eq. 47 is

$$\tau = \mu\left|\frac{dv}{dr}\right| = \frac{\mu r\omega}{c}$$

and the frictional moment for a bearing of length l with surface area $A = 2\pi rl$ becomes

$$M = \tau Ar = \frac{2\pi\mu r^3 l\omega}{c} \tag{52}$$

where μ is the absolute viscosity of the lubricant.

(*d*) *Disk and Pivot Friction.* Friction between circular surfaces under normal pressure is encountered in pivot bearings, clutch plates, and disk brakes. Consider the two flat circular disks of Fig. 79 whose shafts are mounted in bearings (not shown) so that they can be brought into contact under the axial force P. The maximum torque that this clutch can transmit will be equal to the torque M required to slip one disk against the other. If p is the normal pressure at any location between the plates, the frictional force acting on an elemental area is $fp\ dA$, where f is the friction coefficient and dA is the area $r\ dr\ d\theta$ of the element. The moment of this elemental friction force about the shaft axis is $fpr\ dA$, and the total moment is

$$M = \int fpr\ dA$$

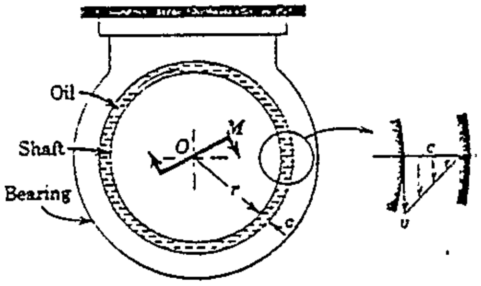

Figure 78

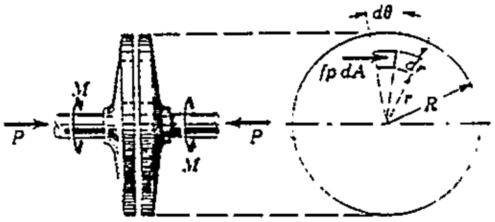

Figure 79

where the integral is evaluated over the area of the disk. To carry out this integral the variation of f and p with r must be known.

In the following examples f is assumed to be constant. Furthermore, if the surfaces are new, flat, and well supported, it is reasonable to assume that the pressure p is constant and uniformly distributed so that $\pi R^2 p = P$. Substituting this constant value of p in the expression for M gives

$$M = \frac{fP}{\pi R^2} \int_0^{2\pi} \int_0^R r^2 \, dr \, d\theta = \tfrac{2}{3} fPR \tag{53}$$

This result may be interpreted as being equivalent to the moment due to a friction force fP acting at a distance $2R/3$ from the center of the shaft.

~~If the friction disks are rings~~, the limits of integration are the inside and outside radii R_i and R_o, respectively, and the frictional torque becomes

$$\boxed{M = \tfrac{2}{3} fP \frac{R_o^3 - R_i^3}{R_o^2 - R_i^2}} \tag{53a}$$

After some wear of the surfaces has taken place, it is found that the frictional moment decreases somewhat. When the wearing-in period is over, the surfaces retain their new relative shape and further wear is therefore constant over the surface. This wear depends on the circumferential distance traveled and the pressure p. Since the distance traveled is proportional to r, the expression $rp = K$ may be written, where K is a constant. The value of K is determined by equating the axial forces to zero, or

$$P = \int p \, dA = K \int_0^{2\pi} \int_0^R dr \, d\theta = 2\pi KR$$

With $pr = K = P/(2\pi R)$, the expression for M may be written

$$M = \int fpr \, dA = \frac{fP}{2\pi R} \int_0^{2\pi} \int_0^R r \, dr \, d\theta$$

which becomes

$$M = \tfrac{1}{2} fPR \tag{54}$$

The frictional moment for worn-in plates is, therefore, only $(\frac{1}{2})/(\frac{2}{3})$, or $\frac{3}{4}$ as much as for new surfaces.

If the friction disks are rings of inside radius R_i and outside radius R_o, substitution of these limits in the integrations shows that the frictional torque for worn-in surfaces is

$$M = \tfrac{1}{2} fP(R_o + R_i) \tag{54a}$$

(*e*) *Belt Friction.* The impending slippage of flexible members such as belts and ropes over sheaves and drums is of importance in the design of belt drives of all types, band brakes, and hoisting rigs. In Fig. 80 is shown a drum subjected to the two belt tensions T_1 and T_2, the torque M necessary to prevent rotation, and a bearing reaction R. With M in the direction shown, T_2

MECHANICAL ENGINEERS' HANDBOOK

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SIXTH EDITION

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McGRAW-HILL BOOK COMPANY

.ence of letters indicates that this member
· is in tension.
· chord there are now two unknown stresses.
second joint. The stress HB is of the same
:n the opposite direction. The stress in the
sion.

oaded with a dead load of 21,760 lb uniformly
.th a wind load of 13,600 lb on the right side.
:ontal components of the supporting forces are
may be determined graphically by first assum-
:e funicular polygon construction (see p. 3-13).
external forces plotted from the results of the
offers a check on this most important part of

.d is (16/34)13,600 = 6,400 lb. The H com-
.o be 6,400/2 = 3,200 lb. The vertical com-
.000 lb. Taking moments about the right end
.) = $60V_1$. $\therefore V_1 = 14{,}733$; $V_2 = 21{,}760 +$
:.000 − 14,733 = 19,027.

The polygon of external forces can now be
nstructed, as in Fig. 8. The dotted part of the
agram is the combination of the dead and wind
.ds, assuming that they are each concentrated.
.e dotted line BK is the resultant of these
.ds. The supporting forces KA and AB are
.termined by plotting to scale their horizontal
d vertical components as calculated. The
lygon of external forces for the truss is
.$DEFGHIJKAB$, and must check with the
lygon shown dotted. The forces GH, HI, IJ,
d JK are the resultants of the forces acting at
: joints on the right side of the truss.

When supporting forces are to be deter-
.n . loads *may* be concentrated at the
es tion of their resultants, but when
erna. forces are to be determined the loads
.st be distributed at the various joints. Start
:h some joint of the truss where there are only
.am. Fig. 6. The magnitudes of the stresses
.y from the lengths of the corresponding lines
agram. The nature of the stress (tension or
.notation.

.d by the truss of Fig. 7. It is impossible
.te the stress diagram, as it will be found
mbers meeting at the left end and at the
.no joint with less than three unknown
.lty, some unknown stress may be calcu-
.eding with the graphical solution.

.is case is that in the middle member of the
at the middle of the upper chord. $16RA$ =
· 15 + 7½)] − (1,360 × 30) = 227,600. ∴
: diagram and proceed.

. noting that T must be on a line through
.gh f parallel to fs and also that TS must
.tionship may be indicated by any points
.g as $T'S'$ is parallel to ts. Furthermore,
. sr. If T' and S' are arbitrarily selected
However, U must lie on a line through V
.d by moving the triangle $T'S'U'$ so that
:he line through V parallel to uv. This
.ermined.

FRICTION

BY

Dudley D. Fuller

REFERENCES: Bowden and Tabor, "The Friction and Lubrication of Solids," Oxford. Fuller, "Theory and Practice of Lubrication for Engineers," Wiley. Ham and Crane, "Mechanics of Machinery," McGraw-Hill. Bevan, "Theory of Machines," Longmans. Vallance and Doughtie, "Design of Machine Members," McGraw-Hill.

Friction is the resistance that is encountered when two solid surfaces slide or tend to slide over each other. The surfaces may be either dry or lubricated. In the first case, when the surfaces are free from contaminating fluids, or films, the resistance is called dry friction. The friction of brake shoes on the rim of a wheel is an example of dry friction.

When the rubbing surfaces are separated from each other by a very thin film of lubricant, the friction is that of [redacted]. The lubrication depends in this case on the strong adhesion of the lubricant to the material of the rubbing surfaces; the layers of lubricant slip over each other instead of the dry surfaces. A journal when starting, reversing, or turning at very low speed under a heavy load is an example of the condition that will cause boundary lubrication. Other examples are gear teeth (especially hypoid gears), cutting tools, wire-drawing dies, power screws, bridge trunions, and the running-in process of most lubricated surfaces.

When the lubrication is arranged so that the rubbing surfaces are separated by a fluid film, and the load on the surfaces is carried entirely by the hydrostatic or hydrodynamic pressure in the film, the friction is that of complete (or viscous) lubrication. In this case, the frictional losses are due solely to the internal fluid friction in the film. Oil ring bearings, bearings with forced feed of oil, pivoted shoe-type thrust bearings operating in an oil bath, hydrostatic oil pads, oil lifts, and step bearings are instances of complete lubrication.

Incomplete lubrication or mixed lubrication takes place when the load on the rubbing surfaces is carried partly by a fluid viscous film and partly by areas of boundary lubrication. The friction is intermediate between that of fluid and boundary lubrication. Incomplete lubrication exists in bearings with drop-feed, waste-packed, or wick-fed lubrication, or on the guides of a crosshead.

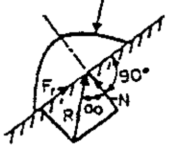

FIG. 1

STATIC AND SLIDING COEFFICIENTS OF FRICTION

In the absence of friction, the resultant of the forces between the surfaces of two bodies pressing upon each other is normal to the surface of contact. With friction, the resultant deviates from the normal.

If one body is pressed against another by a force P, as in Fig. 1, the first body will not move, provided the angle α_0 included between the line of action of the force and a normal to the surfaces in contact does not exceed a certain value which depends upon the nature of the surfaces. The resultant force R has the same magnitude and line of action as the force P. In Fig. 1, R is resolved into two components: a force N normal to the surfaces in contact and a force F_r parallel to the surfaces in contact. From the above statement it follows that

$$F_r \leq N \tan \alpha_0 \leq Nf_0$$

Table 1. Coefficients of Static and Sliding Friction

(Reference letters indicate the lubricant used; numbers in parentheses give the sources. See footnote)

Materials	Static Dry	Static Greasy	Sliding Dry	Sliding Greasy
~~Hard steel on hard steel~~	~~0.78 (1)~~	0.11 (1, a)	~~0.42 (2)~~	0.029 (5, h)
		0.23 (1, b)		0.081 (5, c)
		0.15 (1, c)		0.080 (5, i)
		0.11 (1, d)		0.058 (5, j)
		0.0075 (18, p)		0.084 (5, d)
		0.0052 (18, h)		0.105 (5, k)
				0.096 (5, l)
				0.108 (5, m)
				0.12 (5, a)
Mild steel on mild steel	0.74 (19)		0.57 (3)	0.09 (3, a)
				0.19 (3, u)
Hard steel on graphite	0.21 (1)	0.09 (1, a)		
Hard steel on babbitt (ASTM No. 1)	0.70 (11)	0.23 (1, b)	0.33 (6)	0.16 (1, b)
		0.15 (1, c)		0.06 (1, c)
		0.08 (1, d)		0.11 (1, d)
		0.085 (1, e)		
Hard steel on babbitt (ASTM No. 8)	0.42 (11)	0.17 (1, b)	0.35 (11)	0.14 (1, b)
		0.11 (1, c)		0.065 (1, c)
		0.09 (1, d)		0.07 (1, d)
		0.08 (1, e)		0.08 (11, h)
Hard steel on babbitt (ASTM No. 10)		0.25 (1, b)		0.13 (1, b)
		0.12 (1, c)		0.06 (1, c)
		0.10 (1, d)		0.055 (1, d)
		0.11 (1, e)		
Mild steel on cadmium silver				0.097 (2, f)
Mild steel on phosphor bronze			0.34 (3)	0.173 (2, f)
Mild steel on copper lead				0.145 (2, f)
Mild steel on cast iron		0.183 (15, c)	0.23 (6)	0.133 (2, f)
Mild steel on lead	0.95 (11)	0.5 (1, f)	0.95 (11)	0.3 (11, f)
Nickel on mild steel			0.64 (3)	0.178 (3, x)
Aluminum on mild steel	0.61 (8)		0.47 (3)	
Magnesium on mild steel			0.42 (3)	
Magnesium on magnesium	0.6 (22)	0.08 (22, y)		
Teflon on Teflon	0.04 (22)			0.04 (22, f)
Teflon on steel	0.04 (22)			0.04 (22, f)
Tungsten carbide on tungsten carbide	0.2 (22)	0.12 (22, a)		
Tungsten carbide on steel	0.5 (22)	0.08 (22, a)		
Tungsten carbide on copper	0.35 (23)			
Tungsten carbide on iron	0.8 (23)			
Bonded carbide on copper	0.35 (23)			
Bonded carbide on iron	0.8 (23)			
Cadmium on mild steel			0.46 (3)	
Copper on mild steel	0.53 (8)		0.36 (3)	0.18 (17, a)
Nickel on nickel	1.10 (16)		0.53 (3)	0.12 (3, w)
Brass on mild steel	0.51 (8)		0.44 (6)	
Brass on cast iron			0.30 (6)	
Zinc on cast iron	0.85 (16)		0.21 (7)	
Magnesium on cast iron			0.25 (7)	
Copper on cast iron	1.05 (16)		0.29 (7)	
Tin on cast iron			0.32 (7)	
Lead on cast iron			0.43 (7)	
Aluminum on aluminum	1.05 (16)		1.4 (3)	
Glass on glass	0.94 (8)	0.01 (10, p)	0.40 (3)	0.09 (3, a)
		0.005 (10, q)		0.116 (3, v)
Carbon on glass			0.18 (3)	
Garnet on mild steel			0.39 (3)	
Glass on nickel	0.78 (8)		0.56 (3)	
Copper on glass	0.68 (8)		0.53 (3)	
Cast iron on cast iron	1.10 (16)		0.15 (9)	0.070 (9, d)
				0.064 (9, n)
Bronze on cast iron			0.22 (9)	0.077 (9, n)
Oak on oak (parallel to grain)	0.62 (9)		0.48 (9)	0.164 (9, r)
				0.067 (9, s)
Oak on oak (perpendicular)	0.54 (9)		0.32 (9)	0.072 (9, s)
Leather on oak (parallel)	0.61 (9)		0.52 (9)	
Cast iron on oak			0.49 (9)	0.075 (9, n)
Leather on cast iron			0.56 (9)	0.36 (9, t)
				0.13 (9, n)
Laminated plastic on steel			0.35 (12)	0.05 (12, t)
Fluted rubber bearing on steel				0.05 (13, t)

(1) Campbell, *Trans. ASME*, 1939; (2) Clarke, Lincoln, and Sterrett, *Proc. API*, 1935; (3) Beare and Bowden, *Phil. Trans. Roy. Soc.*, 1935; (4) Dokos, *Trans. ASME*, 1946; (5) Boyd and Robertson, *Trans. ASME*, 1945; (6) Sachs, *Zeit. f. angew. Math. und Mech.*, 1924; (7) Honda and Yamada, *Jour. I of M*, 1925; (8) Tomlinson, *Phil. Mag.*, 1929; (9) Morin, *Acad. Roy. des Sciences*, 1838; (10) Claypoole, *Trans. ASME*, 1943; (11) Tabor, *Jour. Applied Phys.*, 1945; (12) Eyssen, General Discussion on Lubrication, *ASME*, 1937; (13) Brazier and Holland-Bowyer, General Discussion on Lubrication, *ASME*, 1937; (14) Bur-

where $f_s = \tan \alpha_s$ is called the **coefficient of friction of rest (or of static friction)** and α_s is the **angle of friction of rest (or angle of repose).**

If the normal force N between the surfaces is kept constant, and the tangential force F_s is gradually increased, there will be no motion while $F_s < Nf_s$. A state of *impending motion* is reached when F_s nears the value of Nf_s. If one surface slides over the other, being pressed together by a normal force N, a frictional force F resisting the motion must be overcome. This force is usually smaller than F_s. The force F is commonly expressed as $F = fN$, where f is the **coefficient of sliding friction, or kinetic friction.** In the range of practical velocities of sliding, the coefficients of sliding friction are smaller than the coefficients of static friction. With small velocities of sliding and very clean surfaces, the two coefficients do not differ appreciably.

Under moderate pressures, the frictional force is proportional to the normal load on the rubbing surfaces. It is independent of the pressure per unit area of the surfaces. The coefficient of friction is approximately independent of the rubbing speed, when the speed is sufficiently low so as not to affect the temperature of the surface; at higher velocities, the coefficient of friction decreases as the velocity increases.

The coefficients of friction for dry surfaces (dry friction) depend on the materials sliding over each other and on the finished condition of the surfaces. With greasy (boundary) lubrication, the coefficients depend both on the materials and conditions of the surfaces and on the lubricants employed.

Coefficients of friction are sensitive to atmospheric dust and humidity, oxide films, surface finish, velocity of sliding, temperature, vibration, and the extent of contamination. In many instances the degree of contamination is perhaps the most important single variable. For example, in the table below, values for the static coefficient of friction of steel on steel are listed, and, depending upon the degree of contamination of the specimens, the coefficient of friction varies effectively from ∞ (infinity) to 0.013.

Coefficients of Static Friction for Steel on Steel

Test condition	f_s	Ref.*
Degassed at elevated temp in high vacuum	∞ (weld on contact)	20
Grease-free in vacuum	0.78	1
Grease-free in air	0.39	8
Clean and coated with oleic acid	0.11	1
Clean and coated with solution of stearic acid	0.013	21

* See footnote to Table 1.

The most effective lubricants for non-fluid lubrication are generally those which react chemically with the solid surface and form an adhering film that is attached to the surface with a chemical bond. This action depends upon the nature of the lubricant and upon the reactivity of the solid surface. The table below indicates that a fatty acid, such as found in animal, vegetable, and marine oils, reduces the coefficient of friction markedly only if it can react effectively with the solid surface. Paraffin oil is almost completely non-reactive.

Values in Table 1 of sliding and static coefficients have been selected largely from recent investigations where these variables have been very carefully controlled. They

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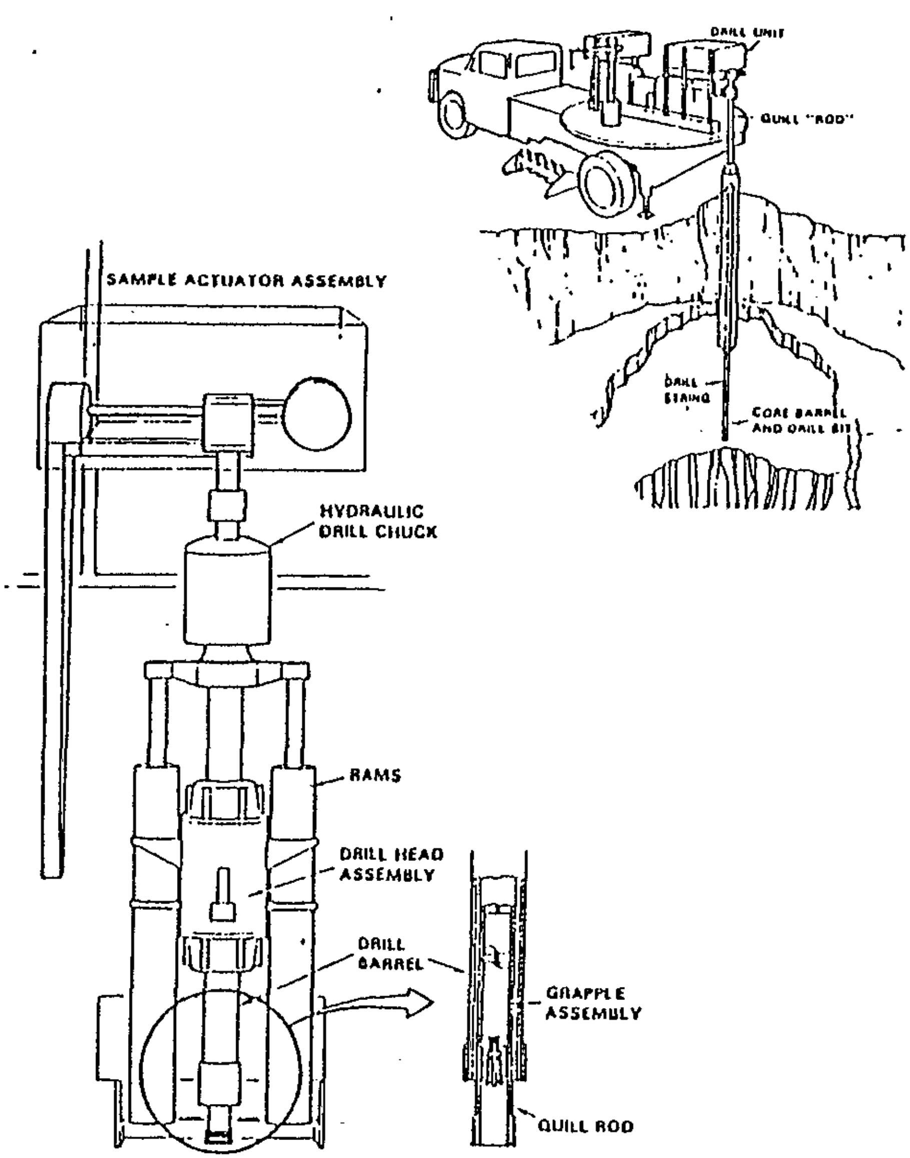
FOOTNOTE, TABLE 1 (cont.)

well, *Jour. SAE*, 1942; (15) Stanton, "Friction," Longmans; (16) Ernst and Merchant, Conference on Friction and Surface Finish, M.I.T., 1940; (17) Gongwer, Conference on Friction and Surface Finish, M.I.T., 1940; (18) Hardy and Bircumshaw, *Proc. Roy. Soc.*, 1925; (19) Hardy and Hardy, *Phil. Mag.*, 1919; (20) Bowden and Young, *Proc. Roy. Soc.*, 1951; (21) Hardy and Doubleday, *Proc. Roy. Soc.*, 1923; (22) Bowden and Tabor, "The Friction and Lubrication of Solids," Oxford; (23) Shooter, *Research*, 4, 1951.

(*a*) Oleic acid; (*b*) Atlantic spindle oil (light mineral); (*c*) castor oil; (*d*) lard oil; (*e*) Atlantic spindle oil plus 2 percent oleic acid; (*f*) medium mineral oil; (*g*) medium mineral oil plus ½ percent oleic acid; (*h*) stearic acid; (*i*) grease (zinc oxide base); (*j*) graphite; (*k*) turbine oil plus 1 percent graphite; (*l*) turbine oil plus 1 percent stearic acid; (*m*) turbine oil (medium mineral); (*n*) olive oil; (*p*) palmitic acid; (*q*) ricinoleic acid; (*r*) dry soap; (*s*) lard; (*t*) water; (*u*) rape oil; (*v*) 3-in-1 oil; (*w*) octyl alcohol; (*x*) triolein; (*y*) 1 percent lauric acid in paraffin oil.


DRILL UNIT
QUILL "ROD"
DRILL STRING
CORE BARREL AND DRILL BIT
SAMPLE ACTUATOR ASSEMBLY
HYDRAULIC DRILL CHUCK
RAMS
DRILL HEAD ASSEMBLY
DRILL BARREL
GRAPPLE ASSEMBLY
QUILL ROD

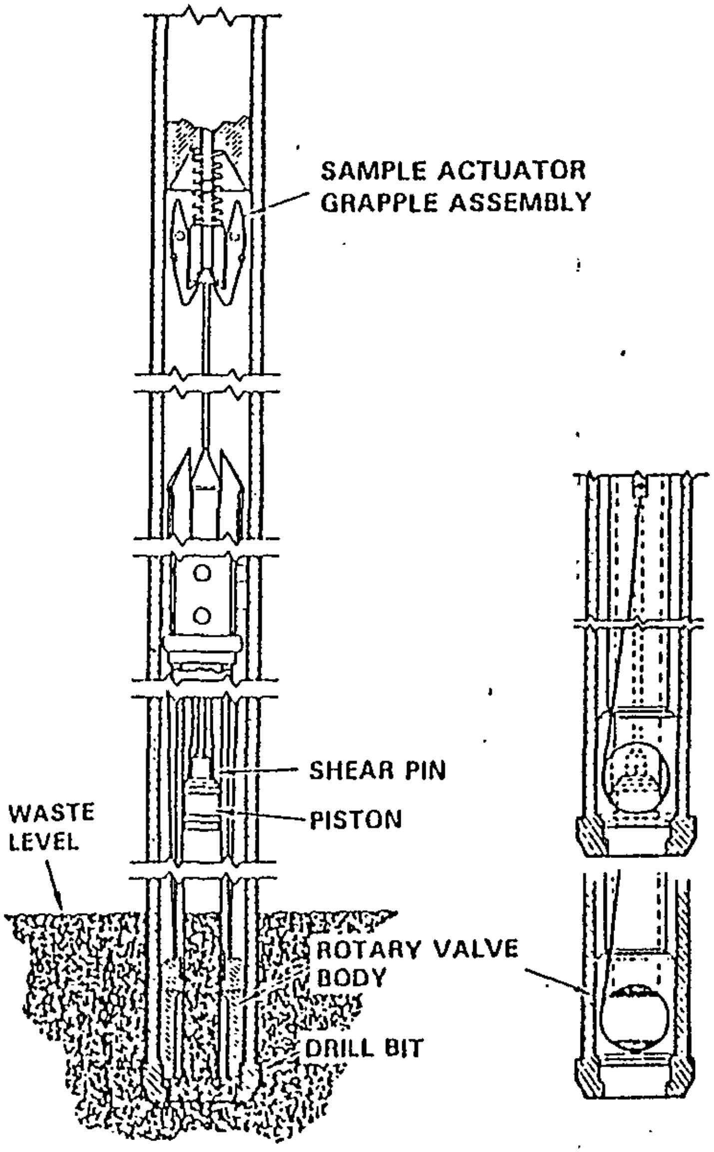

SAMPLE ACTUATOR GRAPPLE ASSEMBLY
SHEAR PIN
PISTON
WASTE LEVEL
ROTARY VALVE BODY
DRILL BIT

TABLE A.1

Based on Tank Temperature of BY-104 (158°F, 70°C)
for 500 rpm Drill Speed and 6 minute Spin Time

Friction Factor		Pressure Applied, psi				
		0	500	1000	1500	2000
Sliding Greasy	Heat Generation KBTU/hr ft^3	504	7312	14120	20928	27736
	Temperature Rise					
	°F	1	14	27	41	54
0.12	°C	.5	7	15	22	30
	Drill Bit Temperature					
	°F	159	172	185	199	212
	°C	71	78	85	93	100
Sliding Dry	Heat Generation KBTU/hr ft^3	1723	24984	48245	71506	94766
	Temperature Rise					
	°F	3	49	94	140	186
0.41	°C	2	27	53	78	103
	Drill Bit Temperature					
	°F	161	207	253	298	344
	°C	71	97	123	148	173
Static Greasy	Heat Generation KBTU/hr ft^3	966	14015	27064	40113	53161
	Temperature Rise					
	°F	2	28	53	79	104
0.23	°C	1	15	30	44	58
	Drill Bit Temperature					
	°F	160	186	211	237	262
	°C	71	85	100	114	128
Static Dry	Heat Generation KBTU/hr ft^3	3279	47531	91783	136035	180288
	Temperature rise					
	°F	6	93	180	267	354
0.78	°C	4	52	100	148	197
	Drill Bit Temperature					
	°F	164	251	338	425	512
	°C	74	122	170	218	267

Temperature rise are based on the results of HEATING5 runs.

TABLE A.2

Based on Tank Temperature of BY-104 (158°F, 70°C)
for 200 rpm Drill Speed and One Minute Spin Time

Friction Factor		Pressure Applied, psi				
		0	500	1000	1500	2000
Sliding Greasy	Heat Generation KBTU/hr ft^3	201	2925	5648	8371	11094
	Temperature Rise					
	°F	.19	2.8	5.4	8.0	11
0.12	°C	.11	1.6	3.0	4.5	5.9
	Drill Bit Temperature					
	°F	158	161	163	166	169
	°C	70	72	73	74	76
Sliding Dry	Heat Generation KBTU/hr ft^3	689	9994	19298	28602	37906
	Temperature Rise					
	°F	0.66	9.6	18	27	36
0.41	°C	0.37	5.3	10	15	20
	Drill Bit Temperature					
	°F	159	168	176	185	194
	°C	70	75	80	85	90
Static Greasy	Heat Generation KBTU/hr ft^3	386	5606	10825	16045	21264
	Temperature Rise					
	°F	.37	5.4	10	15	20
0.23	°C	.21	3.0	5.8	8.5	11
	Drill Bit Temperature					
	°F	158	163	168	173	178
	°C	70	73	76	79	81
Static Dry	Heat Generation KBTU/hr ft^3	1311	19012	36713	54414	72115
	Temperature rise					
	°F	1.3	18	35	52	69
0.78	°C	.7	10	20	29	38
	Drill Bit Temperature					
	°F	159	176	193	210	227
	°C	71	80	90	99	108

Temperature rise are based on the results of HEATING5 runs.

APPENDIX B

Assumptions and Simplifications

Several simplifications and assumptions were made in order to model the sampling of the waste in single-shell tanks using the HEATING5 heat transfer computer code. The following is a list of assumptions and physical data used in this analysis.

1. Heat transfer across air spaces in the tank is by radiation and natural convection.

2. Thermodynamic properties of the concrete in the dome were assumed to be equal to soil properties to simplify modeling. The real concrete properties were used in the tank walls and base. This assumption will make the dome temperature slightly higher.

3. An adiabatic or insulated boundary was placed at a radial distance of 60 feet from the tank center. This assumptions is reasonable and simulates a tank in the middle of a large array of tanks, all generating the same amount of heat.

4. Lower boundary was placed at 200 feet below grade level at a constant 55°F.

5. A forced convection boundary condition was placed on the soil surface, simulation heat transfer to the atmosphere at 70°F with a heat transfer coefficient of 2.0 BTU/hr ft^2°F.

6. The metal of the primary and secondary shells were ignored.

7. The heat transfer up the drill string was included in the modeling.

8. Axisymmetric symmetry is assumed. The two-dimensional cylinder (R, Z coordinates) heat transfer models are defined in Figure B.1. This assumption tends to make the calculations conservative by reducing the surface area through which heat is transferred to the upper and lower boundaries.

9. The waste is assumed to be cylindrical slab of uniform thickness, thermal conductivity, and power density. Actual tanks have layered solids and varying degrees of nonuniformity of thermal properties. The resultant temperatures may be somewhat higher or lower, depending on how the heat-generating material is distributor.

10. Physical properties:

Material	Thermal Conductivity BTU/ft hr ^{O}F	Density lb/ft^3	Specific Heat BTU/lb ^{O}F
Soil	.25	113	.22
Concrete	.54	144	.21
Drill String	30.0	491	.11
Waste	1.0	1.0	22.60 (Volumetric Specific Heat)
Insulating Concrete	.11	62.0	.2

Soil, concrete, and insulating concrete values from Reference 1.
Waste property from Reference 2.
Drill String Properties is for carbon steel.

References

1. RHO-LD-171, October 1981, G. D. Campbell, "Heat Transfer Analysis for In Situ Disposal of Nuclear Waste in Single- and Double-Shell Underground Storage Tanks."

2. IL# 65610-84-118, June 21,1984,from D. C. Riley to K. G. Carothers, "70 BTU/hr Limit Review for Double-Shell Tanks".

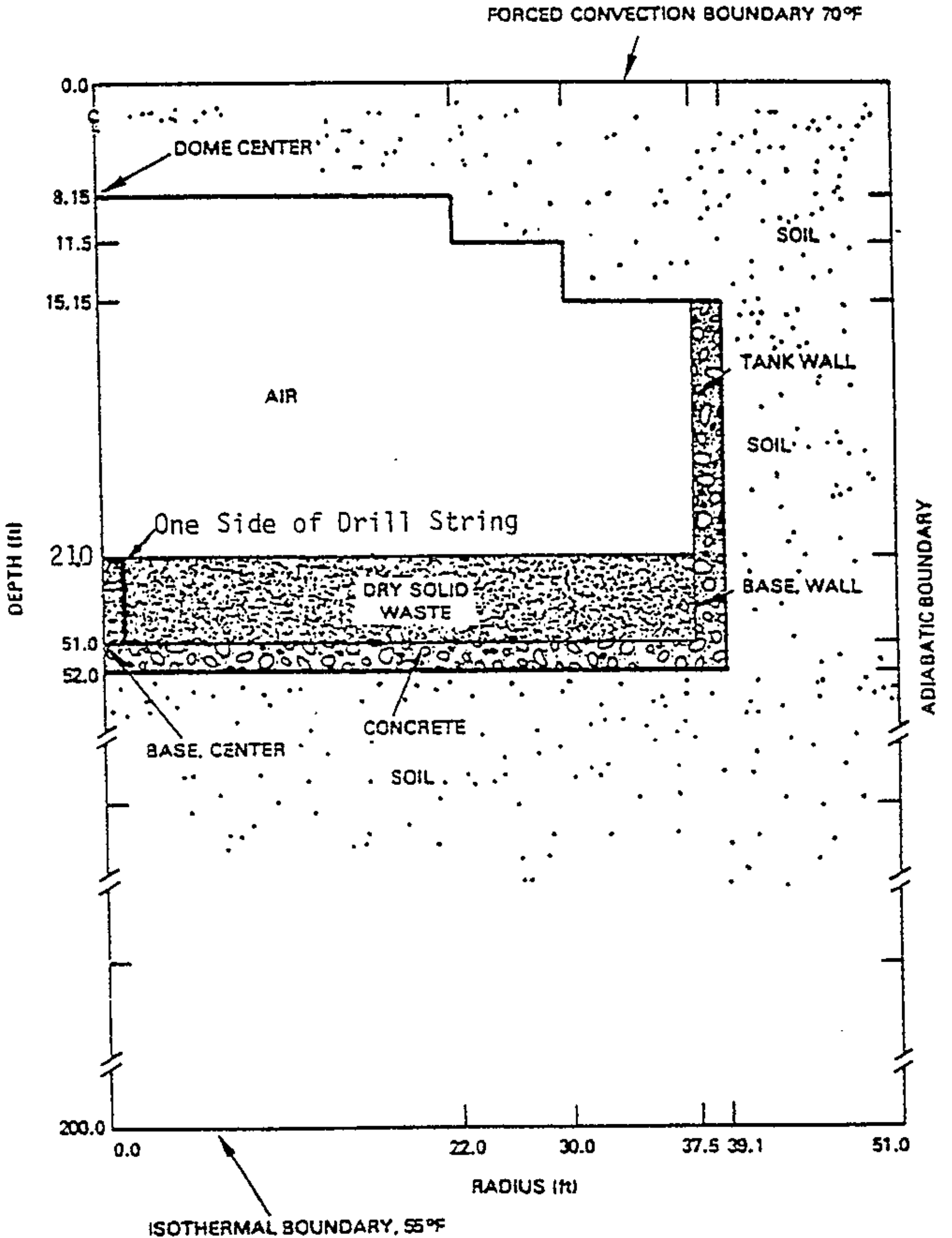


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FIGURE B.1 Single-Shell Tank Heat Transfer Model.